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Type T-130/130-12.8 Cogeneration Steam Turbines with Reheating

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Abstract—The design features and basic thermal scheme of the cogeneration turbines for a rated power capacity of 130 MW and steam parameters of 12.8 MPa and 540/540°C are considered. Figures characterizing the calculated economic effect from the use of steam reheating are presented.

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One of the most important ways in which more efficient cogeneration steam turbines can be developed consists of increasing the parameters of live steam, using a steam reheat system, and constructing power units of a larger capacity [1]. Five design versions of a 250-MW T-250/300-23.5 turbine for steam parameters of 23.5 MPa and 540/540°C of ZAO Ural Turbine Works (UTZ) and two design versions of a 180-MW T-180/210-12.8 turbine for steam parameters of 12.8 MPa and 540/540°C of Leningrad Metal Works (LMZ) are currently in operation at cogeneration stations in Russia and other countries of the former Soviet Union [2]. The majority of cogeneration turbines for live-steam parameters of 12.8 MPa and 555°C and smaller capacity (60–120 MW) that are in operation and still commercially available from turbine manufacturing plants like UTZ are not furnished with a steam reheat system. This paper addresses the problem of developing turbines for a rated capacity of 130 MW equipped with a steam reheat system and having steam parameters of 12.8 MPa and 540/540°C.

It should be pointed out that a boiler suitable for operation with these turbines and well proven by experience gained from its long-term operation has already been available. This is an Ep-270-140-545/545 (PK-38) black coal-fired boiler, the steam at the outlet from which has parameters of 13.8 MPa and 545°C, and that at the steam reheater outlet, 3.1 MPa and 545°C. The boiler has a steam output of 270 t/h. A PK-38-2MPP unit is the boiler's latest model. Intended for operation as part of a power unit comprising two boilers and one turbine (a so-called twin power unit), this once-through steam boiler has a Π-shaped layout of heating surfaces, rests on its own frame, and is installed in a dedicated building.

Due to the losses in the live steam mains, the high-pressure steam before the turbine stop valve has parameters of 12.8 MPa and 540°C, and the steam at the reheat system outlet has a temperature of 540°C.

Below, two design versions of turbines are considered: a two-cylinder T-130/130-12.8-1 turbine (Fig. 1) and a three-cylinder T-130/130-12.8-2 turbine (Fig. 2). The rated and maximum flowrates of live steam to the turbine during its operation in the cogeneration mode are equal to around 470 and 485 t/h.

The two-cylinder turbine has a single-walled cast high-pressure cylinder (HPC). Steam is fed to the HPC from a standalone stop valve through four unbalanced control valves installed on the cylinder's shell. The stop and control valves are fully standardized with the similar valves of a T-110/120-12.8-5M turbine. The HPC comprises a single-row control stage with a mean diameter of 1100 mm and nine pressure stages with rotor-blade roots 800 mm in diameter. The control stage is fully standardized with the similar stage used in the HPC of a T-250/300-23.5 turbine. Pressure stages 3–10 are almost completely standardized in the geometrical parameters of their blade systems with stages 2–9 of a T-110/120-12.8-5M turbine. The second pressure stage is a new one. The rotor blades of the HPC stages are furnished with high-efficient axial-radial shroud seals [3].

The low-pressure cylinder (LPC) has a combined welded and cast design. Its cast steam-admission part is almost fully standardized with the intermediate-pressure cylinder IPC-1 of a T-250/300-23.5 turbine.

The exhaust part and stages 26 and 27 are fully standardized with those of a PT-90/125-12.8/10-2 turbine. The last-stage rotor blades have a height of 660 mm.

The LPC comprises 17 stages. Stages 11–19 have rotor blades with roots 1077 mm in diameter; the disks of these stages are forged together with the shaft. Axial-radial seals are used above the shrouds of these stages. In their profile, the rotor blades are similar to those of stages 15–22 used in the IPC-1 of a T-250 turbine. Stages 19–25 are fully standardized in their geometrical parameters with stages 16–22 of a T-185/220-12.8-2 turbine. The basic thermal scheme of the T-130/130-12.8-1 turbine unit is shown in Fig. 3. The regeneration system includes four low-pressure heaters (LPHs), a deaerator, and three high-pressure heaters (HPHs). The main advantage of this system is that the condensate of LPH-

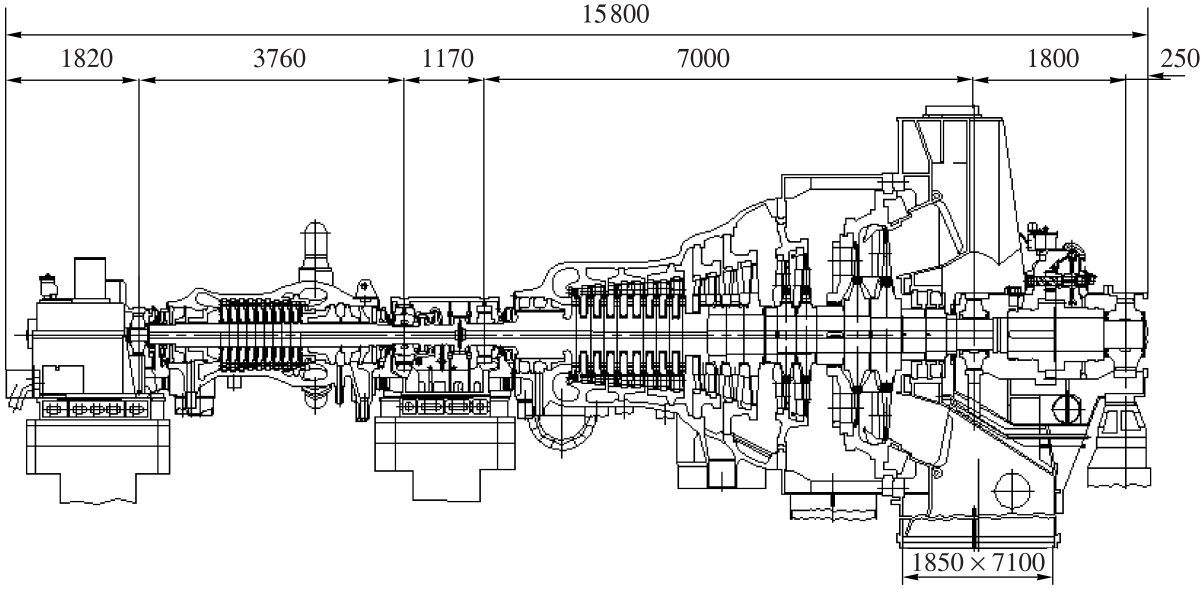

Fig. 1. Type T-130/130-12.8-1 steam turbine.

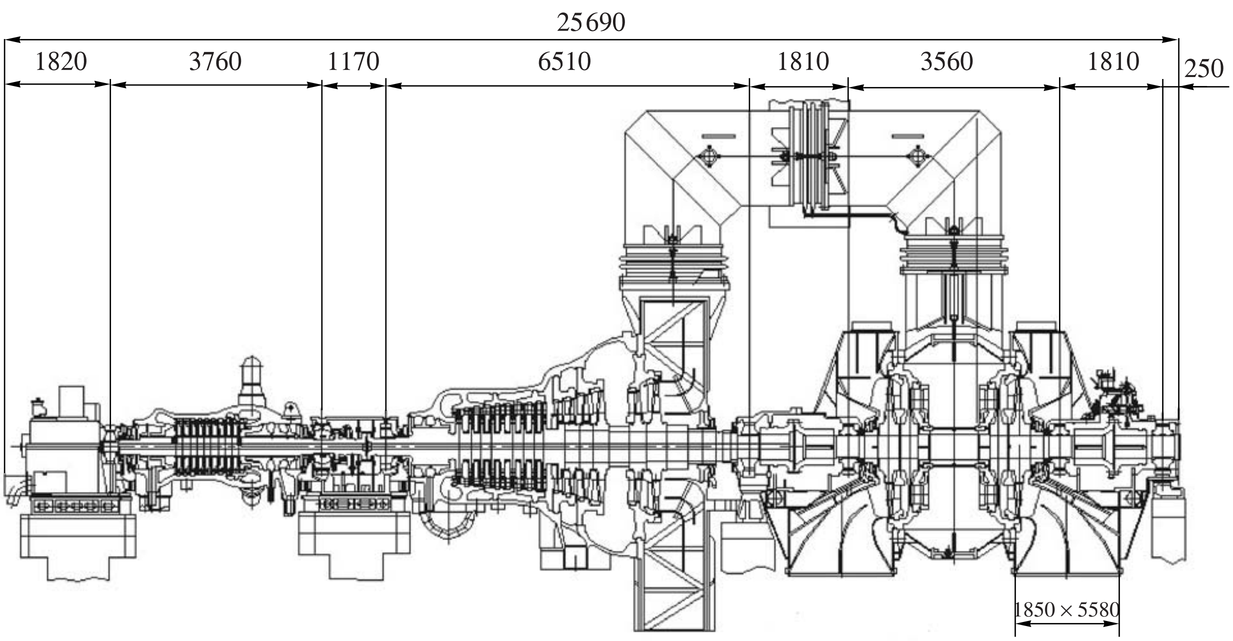

Fig. 2. Type T-130/130-12.8-2 steam turbine.

2 heating steam is drained into the hot well of the horizontal delivery-water heater DWH-2, and the condensate of LPH-1 heating steam, into the hot well of the DWH-1. It should be noted that none of these LPHs are equipped with level regulators. The use of this technical solution makes it possible to do without back-pressure valves on the lines through which steam is supplied to the LPH-1 and LPH-2. The lines through which the condensate of LPH-1 and LPH-2 heating steam is drained into the hot wells are furnished with funnels; this allows these lines also to be made without check valves [4]. The condenser is equipped with a built-in bundle for passing circulation or makeup water. The design does not make a provision for the condenser to operate on the built-in bundle with passing heat-network water. The turbine is equipped with two PSG-2300 delivery-water heaters, each having a heat-transfer surface area of 2300 m^2 and designed for a rated heat-network water flowrate of 4500 m^3/h, and a K-6000 condenser having a heat-transfer surface area

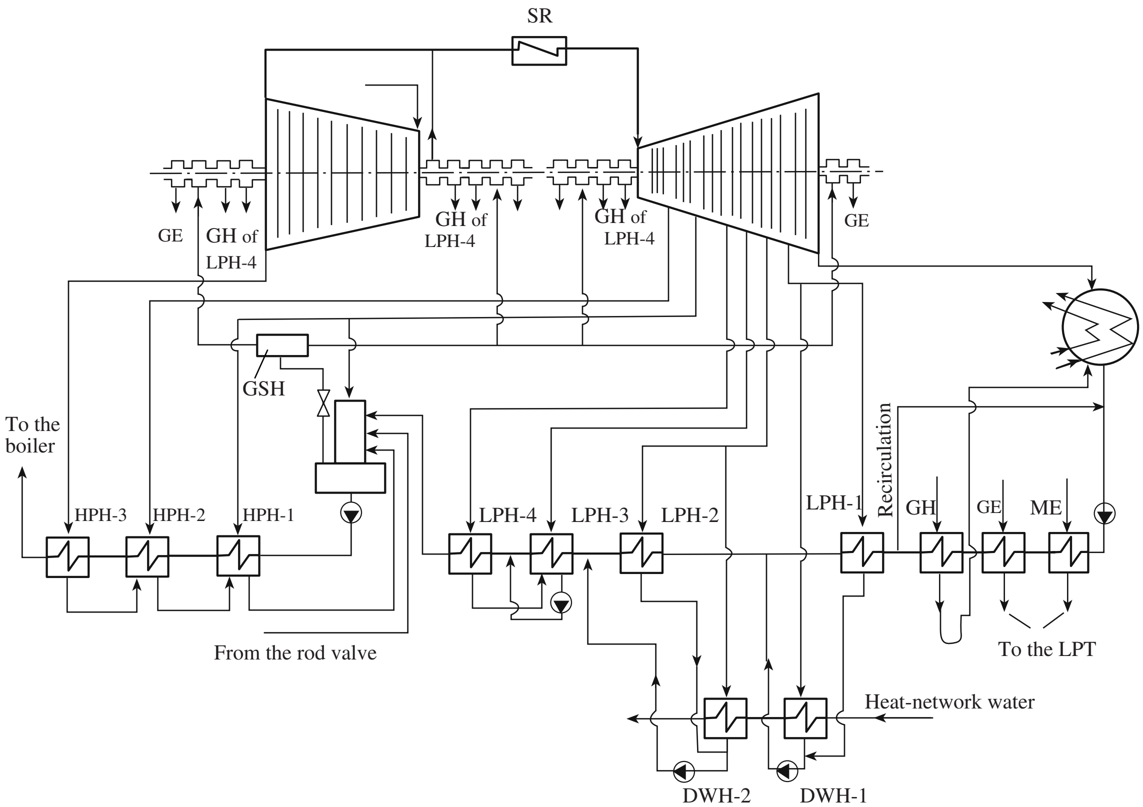

Fig. 3. Basic thermal scheme of the T-130/130-12.8-1 turbine unit. GE—gland ejector, GH—gland heater, GSH—gland steam header, ME—main ejector, SR—steam reheating, and LPT—low-point tank.

of 6000 m^2 and designed for a cooling water flowrate of 13 500 m^3/h.

The three-cylinder T-130/130-12.8-2 turbine incorporates an HPC standardized with that of the T-130/130-12.8-1 turbine. The steam-admission part and stages 11–25 of the IPC of the former are also fully standardized with the corresponding elements of the latter. The turbine's LPC is fully standardized with that of the T-110/120-12.8-5M turbine. The last-stage rotor blades have a height of 550 mm. With the turbine rotor design comprising three cylinders, it became possible to make the second cylinder more compact. The axial distance between the LPC supports of the two-stage turbine is 7000 mm, and that between the IPC supports of the three-stage turbine is 6510 mm. The turbine's basic thermal scheme is almost identical with that of the T-130/130-12.8-1 turbine unit.

The turbine is equipped with two PSG-2300 delivery-water heaters and a K-12000 condenser group having a heat-transfer surface area of 12000 m^2 and designed for a cooling water flowrate of 16000 m^3/h. The condenser group is furnished with built-in bundles for passing circulation, makeup, or heat-network water.

The economic efficiency of the new turbines with steam reheating was assessed in comparison with that of the serially produced T-110/120-12.8-5M turbine (see table).

The indicators characterizing the economic efficiency of the turbines were calculated for the operation with the use of a two-stage delivery-water heating arrangement at a pressure in the upper heating extraction of 0.098 MPa and a return heat-network water temperature of 50°C, parameters close to those established for the average winter operating conditions in the European part of Russia.

The presented data on the electric capacity, thermal load, specific heat rate during operation in condensing modes, and flowrate of steam to the condenser during operation in cogeneration modes were obtained from a calculation of heat balances that took into account the effect of each turbine's design: the efficiency of flow path compartments, steam leakage through the end seals and valve rods, mechanical losses and losses in the generator, pressure drops in the HPC steam admission members and in the steam reheat system, as well as the effect of the condenser. The pressure drops in the first-stage steam reheat path were taken equal to 9%; those in the HPC steam admission members, 3.5%; and those in the second-stage steam reheat path, 3.0%. The rise of temperature in the feedwater pump was taken equal to 7°C. This allows us to consider the obtained data on the efficiency of the use of steam reheating as unbiased.

Table

Parameter	T-110/120-12.8-5M		T-130/130-12.8-1		T-130/130-12.8-2	
	cog	cnd	cog	cnd	cog	cnd
Live steam pressure, MPa	12.8	12.8	12.8	12.8	12.8	12.8
Live steam temperature, °C	555	555	540	540	540	540
Temperature downstream of steam reheater, °C	–	–	540	540	540	540
Steam flowrate to the turbine, G_t, t/h	470	437	466.7	388.3	467.6	386.8
Cooling water temperature, °C	20	27	20	27	20	27
Steam flowrate to the condenser, t/h	10.0	310.7	15.0	287.0	10.0	285.1
Steam pressure in the condenser, kPa	3.9	7.7	3.9	7.95	3.9	7.3
Heat load Q, GJ/h	733	–	808	–	823	–
Electrical capacity, MW	110	120	130	130	130	130
Specific steam rate, kg/(kW h)	4.27	3.64	3.59	2.99	3.60	2.98
Specific heat rate q, kJ/(kW h)	–	9086	–	8674	–	8645
Specific generation of electric energy E, (kW h)/GJ	147.3	–	153.3	–	152.8	–
ε, %	0	0	$\frac{1.84}{2.45}$	4.54	$\frac{1.69}{2.26}$	4.86

Note: cog denotes rated cogeneration operating mode, and cnd denotes the condensing operating mode; the numbers in the numerator are for b_{CPS} = 325 g/(kW · h), and those in the denominator, for b_{CPS} = 380 g/(kW · h).

The relative saving of fuel during operation in condensing modes as compared with its consumption in case of using a T-110/120-12.8-5M turbine was calculated from the formula

$$\varepsilon_{cnd} = \frac{q_1 - q_i}{q_1} \times 100, \quad (1)$$

where q_1 is the specific heat rate for a T-110/120-12.8-5M turbine and q_i is the specific heat rate for the corresponding turbine with steam reheating.

The relative savings of fuel during operation in a cogeneration mode was determined in accordance with [1] as follows:

$$\varepsilon_{cog} = \frac{(E_i - E_1)Q(b_{CPS} - b_{cog})}{B_1}, \quad (2)$$

where E_i is the specific electric energy generated by the corresponding turbine with steam reheating in the cogeneration mode, E_1 is the specific electric energy generated by the T-110/120-12.8-5M turbine in the cogeneration mode, b_{CPS} is the specific fuel rate at the substituting condensing power station, b_{cog} is the specific fuel rate of the cogeneration turbine during its operation in accordance with a heat load curve, and B_1 is the hourly consumption of fuel for the T–110/120–12.8-5M turbine.

The value of E was calculated using the following formula from [1]:

$$E = \frac{\sum_{1}^{z} Gh_i - 3600\Delta N_{mg}}{3600[G_2(h_{ext2} - h_{c2}) + G_z(h_{ext1} - h_{c1})]}, \quad (3)$$

where z is the number of the compartment before the point at which steam is extracted to the DWH-1, G and h_i are the steam flowrate and the utilized drop of heat till the corresponding stage compartment, ΔN_{mg} is the sum of mechanical losses and losses in the generator, G_2 is the flowrate of steam to the DWH-2, h_{ext2} and h_{c2} are the enthalpies of steam in the chamber from which steam is extracted to the DWH-2 and of DHW-2 steam condensate, G_z is the flowrate of steam in the section before the point at which steam is extracted to the DWH-1, and h_{ext1} and h_{c1} are the enthalpies of steam in the chamber from which steam is extracted to the DWH-1 and of DHW-1 steam condensate.

In our analysis, we considered a gas-fired condensing power station equipped with a K-300-23.5 turbine, for which b_{CPS} = 325 g/(kW · h) according to the data in [5], and a power station equipped with a T-110/120-12.8-5M turbine running in a condensing mode, for which b_{CPS} = 380 g/(kW · h). The value b_{cog} = 160 g/(kW · h) was taken from the data of [6] for tur-

bines having live-steam parameters of 12.8 MPa and 555°C and operating in accordance with a heat load curve. The value of B_1 was taken equal to 39700 kg of coal equivalent per hour.

It can be seen from the table that the two-cylinder turbine unit is more economically efficient than the three-cylinder one when it operates in the cogeneration mode (1.84–2.45 versus 1.69–2.26%) and is somewhat inferior to the latter during operation in the condensing mode (4.54 versus 4.86%).

However, since the turbine operates for the most part of the year in cogeneration modes and since the three-cylinder turbine is more expensive than the two-cylinder one and has larger overall dimensions, preference should be given to the two-cylinder turbine.

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